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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 409

THE ELIMINATION OF FIRE HAZARD DUE TO BACK FIRES

By THEODORE THEODORSEN and IRA M. FREEMAN



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AERONAUTICAL SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Symbol	Unit	Symbol
Length-----	l	meter-----	m	foot (or mile)-----	ft. (or mi.)
Time-----	t	second-----	s	second (or hour)-----	sec. (or hr.)
Force-----	F	weight of one kilogram----	kg	weight of one pound-----	lb.
Power-----	P	kg/m/s-----		horsepower-----	hp
Speed-----		km/h-----	k. p. h.	mi./hr.-----	m. p. h.
		m/s-----	m. p. s.	ft./sec.-----	f. p. s.

2. GENERAL SYMBOLS, ETC.

W , Weight = mg	mk^2 , Moment of inertia (indicate axis of the radius of gyration k , by proper subscript).
g , Standard acceleration of gravity = 9.80665 m/s ² = 32.1740 ft./sec. ²	
m , Mass = $\frac{W}{g}$	S , Area.
ρ , Density (mass per unit volume).	S_w , Wing area, etc.
Standard density of dry air, 0.12497 (kg-m ⁻⁴ s ²) at 15° C. and 760 mm = 0.002378 (lb.-ft. ⁻⁴ sec. ²).	G , Gap.
Specific weight of "standard" air, 1.2255 kg/m ³ = 0.07651 lb./ft. ³ .	b , Span.
	c , Chord.
	b^2
	\bar{S} , Aspect ratio.
	μ , Coefficient of viscosity.

3. AERODYNAMICAL SYMBOLS

V , True air speed.	Q , Resultant moment.
q , Dynamic (or impact) pressure = $\frac{1}{2}\rho V^2$.	Ω , Resultant angular velocity.
L , Lift, absolute coefficient $C_L = \frac{L}{qS}$	$\frac{Vl}{\mu}$, Reynolds Number, where l is a linear dimension.
D , Drag, absolute coefficient $C_D = \frac{D}{qS}$	e. g., for a model airfoil 3 in. chord, 100 mi./hr. normal pressure, at 15° C., the corresponding number is 234,000;
D_o , Profile drag, absolute coefficient $C_{D_o} = \frac{D_o}{qS}$	or for a model of 10 cm chord 40 m/s, the corresponding number is 274,000.
D_i , Induced drag, absolute coefficient $C_{D_i} = \frac{D_i}{qS}$	C_p , Center of pressure coefficient (ratio of distance of <i>c. p.</i> from leading edge to chord length).
D_p , Parasite drag, absolute coefficient $C_{D_p} = \frac{D_p}{qS}$	α , Angle of attack.
C , Cross-wind force, absolute coefficient $C_c = \frac{C}{qS}$	ϵ , Angle of downwash.
R , Resultant force.	α_o , Angle of attack, infinite aspect ratio.
i_w , Angle of setting of wings (relative to thrust line).	α_i , Angle of attack, induced.
i_u , Angle of stabilizer setting (relative to thrust line).	α_a , Angle of attack, absolute. (Measured from zero lift position.)
	γ , Flight path angle.

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Langley Memorial Aeronautical Laboratory

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

NAVY BUILDING, WASHINGTON, D. C.

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SUMMARY

A critical study was made of the operation of a type of back-fire arrestor used to reduce the fire hazard in aircraft engines.

A flame arrestor consisting of a pack or plug of alternate flat and corrugated plates of thin metal was installed in the intake pipe of a gasoline engine; an auxiliary spark plug inserted in the intake manifold permitted the production of artificial back fires at will.

It was found possible to design a plug which prevented all back fires from reaching the carburetor.

Analysis of the heat-transfer phenomena in the arrestor shows that (a) the length-diameter ratio of the individual passages is of first importance in determining the effectiveness of the device, (b) that the plug need not be heavy, and (c) that the thermal conductivity of the material of which it is made is only of secondary influence.

Measurements of the pressure drop across the adopted model show that the increase of engine-pumping loss caused by the presence of the arrestor is quite negligible; the reduction in volumetric efficiency amounts to a few per cent.

INTRODUCTION

The reduction of fire hazard in aircraft is one of the most important problems in aeronautics to-day. As long as the carburetor engine, employing highly inflammable fuels, continues to be the source of power for aircraft, considerable risk of fire with possible attendant destruction of property and loss of life will exist. Indeed, the fire danger is held to be one of the chief obstacles confronting the development of aviation. (Reference 1.)

One of the most frequent causes of fire in aircraft is back-firing from the engine through the intake system. This back-firing may occur while the airplane is in flight or as a result of a crash. The chief causes of back fires during flight are (a) intake-valve failure—the valve breaks or fails to seat properly, and (b) change of quality of the fuel mixture—this may result from a defect of the carburetor or from a sudden change of engine speed caused by mechanical failure or other mishaps. (Reference 2.) Brunat (reference 3), basing

his statement on a study of French aeronautical accident statistics, points out that 28 per cent of the fires occurring in flight are definitely attributable to back-firing to the carburetor.

It is probable, too, that many of the fires resulting from crashes are started by flames emerging from the carburetor air intake, the back-firing being the result of the sudden stopping of the engine. In a violent crash the fuel tanks are likely to burst, throwing their contents over the engine compartment.

There is much reason to believe that the two principal sources of aircraft fires are (a) flames from the carburetor, and (b) the hot gases or metal parts of the exhaust system. Any device, then, which succeeds in eliminating the carburetor flame will accomplish much toward solving the general problem of the reduction of fire hazard in aircraft.

Several methods for preventing or confining intake back fires have been proposed. One scheme involves the production of an over-rich mixture at the carburetor with the admission of enough air at a point farther along the intake pipe to give a normal mixture. If a flame were to strike back from the intake valve, it would meet the rich mixture, which would burn slowly. Another suggestion embodies a nonreturn valve in the intake pipe. Neither of these methods have been found to be practicable. (Reference 4.)

A third method employs multiple wire gauze sheets in the carburetor air intake. The action of these screens, based on the principle of the Davy safety lamp, is to remove heat from the flame striking from within the carburetor. A flame originating in the intake manifold is thus restricted to the intake system. Tests of such devices installed on marine engines (reference 5) have demonstrated the possibility of confining all but a very small percentage of back fires by this means.

The work described in this report is an investigation of the feasibility of arresting the flame in the intake pipe *before* it reaches the carburetor. It must be borne in mind that the above-mentioned arrestor on the air inlet will not prevent the ignition of the fuel contained in the carburetor.

EXPERIMENTAL METHOD AND RESULTS

To study the performance of flame arrestors, a special intake pipe was constructed. (Fig. 1.) This pipe was a 10-inch section, 3 inches in diameter, fitted with a spark plug for firing the mixture in the manifold at will. The flame-arresting devices to be tested were mounted in the pipe between two sight holes. The spark-plug end of the pipe was connected to a single-cylinder test engine; the other end was connected to the carburetor. A special timer mounted on the engine camshaft energized the spark plug during every alternate intake stroke. A thermocouple connected to an indicating pyrometer was inserted in the device to be tested.

Tests on wire gauze.—Tests on the effect of gauze screens on the propagation of flames in pipes were conducted by the National Board of Fire Underwriters.

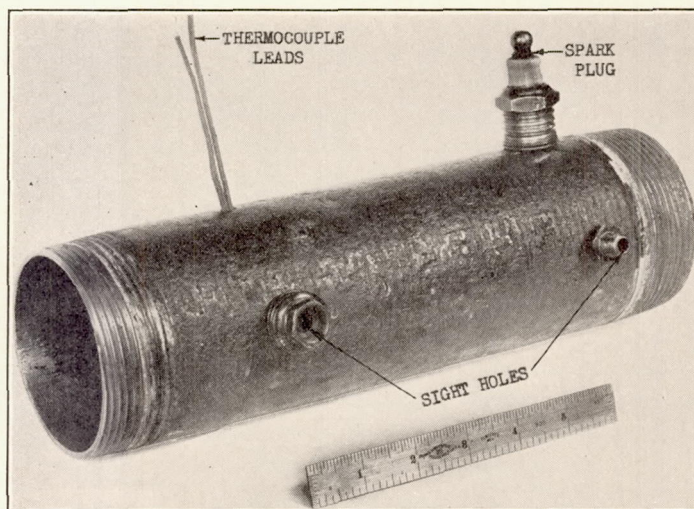


FIGURE 1.—Experimental intake pipe

(Reference 6.) It was found that flames propagated in combustible vapors could be stopped by double cone-shaped arrestors of 40-mesh wire gauze. For pipes under 4 inches in diameter flat screens were found adequate.

A double-walled cone of 20-mesh iron-wire gauze, as well as a triple flat screen of the same material, was tried. In both cases the flame succeeded in passing the screens almost immediately, and a flame could be made to flash out from the air intake opening of the carburetor.

A number of such tests led to the conclusion that a few layers of wire gauze would, in general, be inadequate as intake manifold flame arrestors in engines. The reason is probably that, because of its very short path through the metal screen, the violent pulse of flame succeeds in driving through the screen before the wires are able to conduct away sufficient heat to bring the charge below its ignition point. (Reference 7.) The use of finer mesh screens, as in the underwriters' tests, or the installation of a greater number of

layers of gauze, would involve an increased flow loss in the intake system.

The plate-type flame arrestor.—If the flat screens are replaced by thin plates extending parallel to the axis of the pipe, the cooling of the flame by contact with metal will be increased, because the path of travel of the flame in the arrestor will be lengthened. Moreover, the flow characteristics of this type of arrestor are to be preferred to those of screens, for the resistance of a channel of constant cross section must obviously be smaller than that of any other arrangement. Plate-type arrestors have been described (reference 8), but no test results either as to effectiveness in confining back fires or to influence on engine efficiency have been reported.

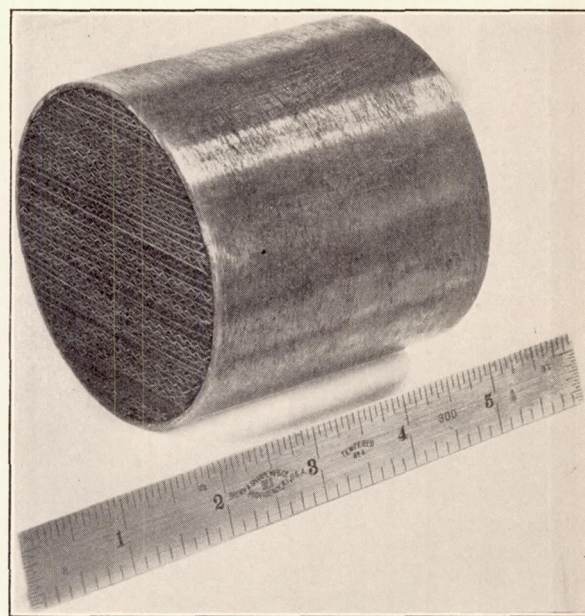


FIGURE 2.—The plate-type flame arrestor

Accordingly, a plate-type flame arrestor was constructed. A plug of alternate corrugated and flat steel plates was built up in a metal sleeve. (Fig. 2.) A dimensioned section of this plug is shown in Figure 3 (a). The weight of the device, when made of steel, is about 3.1 pounds (1.4 kg). This type of construction is simple and insures passages of uniform section throughout.

The plug was inserted into the special intake pipe described above and tested repeatedly at engine speeds varying from 750 r. p. m. to 1,800 r. p. m. The mixture between the arrestor and the intake valve could be made to explode at an average rate of about two to three times per second, and the explosions could be detected by a flame emerging from the sight hole located on the engine side of the plug. The back-firing was allowed to continue for 20 to 30 minutes. During the first 5 to 10 minutes the temperature of the plug, measured at the end nearer the carburetor, attained a final equilibrium value of 200° F. (90° C.). A flame

could under no circumstances be made to penetrate the plug.

Tests with modified plate-type arrestors.—Before investigating the effect of this appliance on the performance of the engine to which it was attached two new plugs were constructed to determine to what extent the openings could be enlarged, or how much the axial length could be diminished, without reducing the effectiveness of the device. The first made was of the same cross section as the 3-inch plug mentioned above (see fig. 3 (a)), but had a length along the axis of only 1 inch. Back-fire tests were conducted as in the previous case, with the following results:

The flame succeeded in passing through the device after about 50 to 100 explosions, starting with the plug cold. The temperature of the plates, measured at the end nearer the carburetor, attained a value of about 350° F. (175° C.), when the charge on the carburetor side was exploded. This temperature at which the 1-inch flame arrestor became ineffective was rather critical, and could be reproduced to within about 40° F.

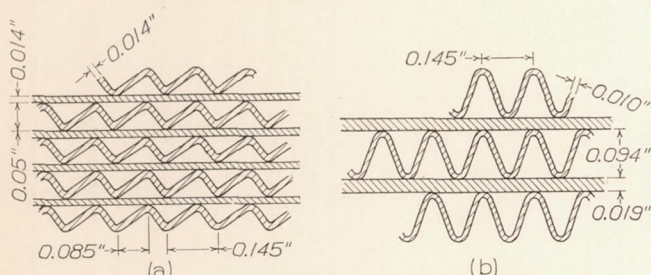


FIGURE 3.—Cross sections showing construction of flame arrestors

After once firing through the plug, the flame would penetrate at the rate of about once for every 20 explosions on the engine side.

The second modification had a depth of 3 inches, but was provided with openings of hydraulic radius about 70 per cent greater than that of the previous models. (Fig. 3(b).) In the back-fire tests the flame penetrated the plug almost immediately. This model was therefore abandoned.

Effect of flame arrestor on engine performance.—In view of the successful operation of the original 3-inch plug, its direct effect on the engine performance was determined. It was evident that the plug must increase the pumping loss of the engine by virtue of its resistance to air flow in the intake pipe, in consequence of which there will be also a decrease in volumetric efficiency. It remains to determine whether these losses are of such magnitude as to make the installation of the flame arrestor objectionable. The plug should be designed to give a minimum air resistance so far as this can be attained without sacrificing its flame-quenching properties.

The pressure drop across the plug was determined for a large range of velocities, using an experimental arrangement shown schematically in Figure 4. The

arrestor was fitted into place in the long pipe, with water manometers connected near each end of the plug to measure the pressure drop across it, and a sharp-edged orifice was employed to measure the air flow. (Reference 9.) Air was drawn through the assembly by a vane-type supercharger connected to the other end of the pipe. Results of the measurements are shown graphically in Figure 5.

Calculations based on the fact that the pumping loss occasioned by the presence of the plug is equal to that expended in forcing the air through the resistance show that this loss is of the order of only one-tenth of 1 per cent of the engine power. Fuel-consumption tests, run with and without the plug, showed no measurable difference.

When the test engine was operated with the plug installed, there was a decrease in brake horsepower amounting to about 5 per cent of the total power at the middle of the speed range of 1,000 to 1,800 r. p. m. As the specific fuel consumption did not change, the entire loss is to be ascribed to a decrease in volumetric efficiency.

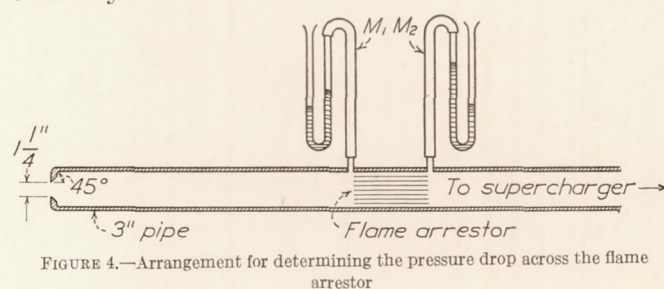


FIGURE 4.—Arrangement for determining the pressure drop across the flame arrestor

It is to be remembered that a single-cylinder engine was used in these tests. A multicylinder engine would have smoother air flow, and the reduction in volumetric efficiency would be considerably less.

The test engine operated at about 35 horsepower; as the arrestors were about 3 inches in diameter, this corresponds to a cross section of about 0.2 square inch per brake horsepower. To reduce the power loss, a plug of greater cross section, inserted into an enlargement of the intake pipe, may be used.

Nature of the flow through the flame arrestor.—It remains to examine the nature of the air flow through the plug. The form of the experimental curve of Figure 5 suggests that the resistance increases with some power of the velocity slightly greater than one. Formulas for viscous fluid flow through tubes of arbitrary cross section have not been developed, but a fair idea of the flow may be obtained by assuming each passage to be replaced by a circular tube of radius comparable to the hydraulic radius of the actual opening. In the present instance this radius was taken to be 0.0225 inch (0.572 mm). The Reynolds Number at the highest velocity measured is then about 1,000, which is below the critical value, and hence the flow through the channels may be expected to be laminar.

The other factor contributing to the resistance of the plug is the effect of the sudden reduction and enlargement of the free area at the entrance and exit of each passage. The actual flow condition is probably somewhat as shown in Figure 6; the rectangular shaded areas represent sections of the plates bounding a single channel.

This end effect may be approximately calculated by semiempirical formulas. (Reference 10.) The effect of viscous drag is calculated by means of the Poiseuille formula. The two contributions are plotted in Figure 5 and the total is given by the dotted line, which agrees well with the experimental curve. The fact that the experimental points lie on a curve which departs but

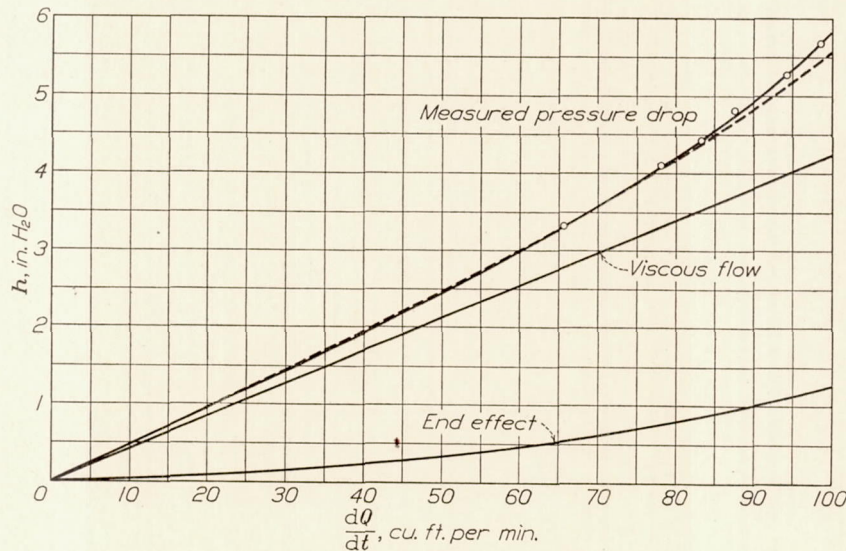


FIGURE 5.—Pressure drop across flame arrestor

$$h, \text{ in. H}_2\text{O} = 0.0510 v + 0.000173 v^2$$

$$(v = \text{velocity in passages, ft./sec.})$$

(In range covered by experimental points, h varies as $sign \propto v^{1.2}$ approx.)

slightly from the linear shows that the flow must be of a nonturbulent nature throughout the range investigated. The end effect is much smaller than the viscous resistance, so that the total resistance of any plug of moderate length is approximately proportional to its length.

ANALYSIS

Heat transfer in the flame arrestor.—A quantitative treatment of the flame-quenching action of a plug of the kind considered is a matter of some difficulty. Nevertheless, it is possible to obtain some knowledge of the principal factors by considering the heat transfer occurring in the pack. The hot gases (products of the explosion) are forced through the plug with great velocity, giving up heat to the metal as they pass through the channels. If these gases are sufficiently cooled before emerging from the plug, the mixture on the carburetor side will not be ignited. Starting with the plug cold, an equilibrium temperature is ultimately attained at which the rate of heat absorbed from the hot gases is equal to that removed by the passage of fresh charges of cold air from the carburetor.

Before turning to a detailed consideration of the heat-transfer phenomena in the plug, the change in temperature of the metal may be calculated approximately by assuming that all the heat contained in the gases as a result of the explosions is delivered to the metal pack. The result indicates a temperature rise of the plug of the order of but 0.5°C. per explosion; that is, the plug possesses a heat capacity which is large compared with that of a charge of gas.

The cooling of the gas by contact with the metal will now be considered. Represent by—

α —heat transfer coefficient of the metal-gas boundary, in $\text{k-cal m}^{-2} \text{h}^{-1} ^\circ\text{C}^{-1}$.

T_w —temperature of the wall at any point, in $^\circ\text{C.}$

T_i —temperature of the gas before entering the plug, in $^\circ\text{C.}$

T —temperature of the gas in $^\circ\text{C.}$ at a distance x cm from the point where it enters the passage.

A —cross section area of one passage in m^2 .

P —perimeter of the cross section of one passage in m.

D —diameter of equivalent circular tube in m.

V —velocity of explosive pulse in plug in m h^{-1} .

C_p —specific heat of gas at constant pressure in $\text{k-cal kg}^{-1} ^\circ\text{C}^{-1}$.

λ —mean thermal conductivity coefficient of the gas in $\text{k-cal m}^{-1} \text{h}^{-1} ^\circ\text{C}^{-1}$.

ρ —density of the gas in kg m^{-3} .

Consider a transverse section, thickness Δx , of the cylindrical passage. As

the gas in this section moves down the tube it loses heat to the walls, and its temperature drops. Equating the heat absorbed by the walls to that removed from the hot gas,

$$\alpha P \Delta x (T - T_w) dt = -\rho A \Delta x C_p dT$$

or

$$dt = -\frac{\rho C_p A}{\alpha P} \cdot \frac{dT}{T - T_w}$$

and the distance advanced in order to reduce the gas temperature by dT is

$$dx = V dt = -NV \frac{dT}{T - T_w} \quad (1)$$

where

$$N = \frac{\rho C_p A}{\alpha P}$$

Nusselt (Hütte, loc. cit. I. S. 402) has developed an expression for the heat transfer in metal tubes

$$\alpha = 0.0255 \frac{\lambda}{D} \left(\frac{V \rho C_p D}{\lambda} \right)^{0.786} \quad (2)$$

The use of this formula, which holds for turbulent flow, is justified here, since the probable high velocity

and the internal disturbances of the back-fire gases assure a condition of turbulence.

Then, using (2),

$$NV = 9.80 D \left(\frac{V \rho C_p D}{\lambda} \right)^{0.214} \quad (3)$$

The velocity V of the hot gases through the plug must be expected to vary greatly throughout the duration of one explosion. It is necessary to apply equation (3) to the *peak* value of this velocity. The duration of each explosion is of no particular significance, because, as already noted, the temperature rise per explosion is negligible. The peak velocity is evidently a function of the combustion velocity of the mixture or rather of the volume of the mixture burning per unit of time. The pressure builds up until the volume discharged through the plug equals the volume-expansion due to the combustion.

On the basis of an assumed rate of combustion and with the flow resistance of the plug known, the velocity through the plug may be estimated with some accuracy. Fortunately, however, a precise determination of this velocity is not of much concern. It will be noticed from equation (3) that the quantity NV , which de-

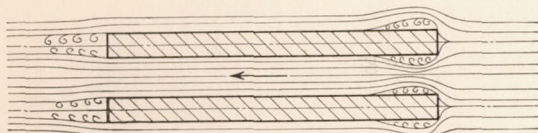


FIGURE 6.—Flow with sudden change of cross section

termines the rate of the temperature drop through the plug, depends very slightly on the velocity, the value of NV being proportional to $V^{0.214}$. Thus, the velocity V may be changed by several hundred per cent without altering the temperature in the plug to any appreciable extent.

The following calculations have been made using a set of arbitrarily chosen numerical values in order to gain some insight into the actual conditions. It is assumed that no combustion occurs after the gases enter the plug. This assumption is, of course, far from being representative of actual conditions. The quantity NV is not given correctly by equation (3), but is in all probability considerably greater because of combustion in the plug proper. It must consequently be borne in mind when employing equation (3) that the result is too favorable.

Uniform wall temperature.—The rate at which the temperature of the gas decreases as it passes through the pack may be calculated from (1). It is, however, first necessary to assume a certain variation of T_w , the temperature of the wall at any point, with the distance x from the front of the pack. Consider first the case where the axial temperature gradient in the plug proper is zero. This condition implies the use of a material of infinite conductivity. Here T_w is con-

stant, and the integration of (1) gives, after imposing the condition $T = T_i$ when $x = 0$,

$$T = T_w + (T_i - T_w)e^{-x/NV} \quad (4)$$

Taking the initial gas temperature $T_i = 600^\circ \text{C}$. and the equilibrium wall temperature $T_w = 165^\circ \text{C}$., and using a value of $NV = 4 \text{ cm}$ (4) becomes

$$T = 165 + 435e^{-0.25x} \quad (5)$$

The result is shown graphically as curve a in Figure 7. The gas experiences a rapid decrease in temperature as it passes through the arrestor after traveling a distance of about 4.75 cm its temperature is reduced from 600°C . to 300°C . and the exit temperature ($x = 7.5 \text{ cm}$) is about 230°C .

For the coarser 3-inch plug, the value of NV turns out to be about 7.5 cm, and this gives, by equation (4), an exit gas temperature of about 325°C . This

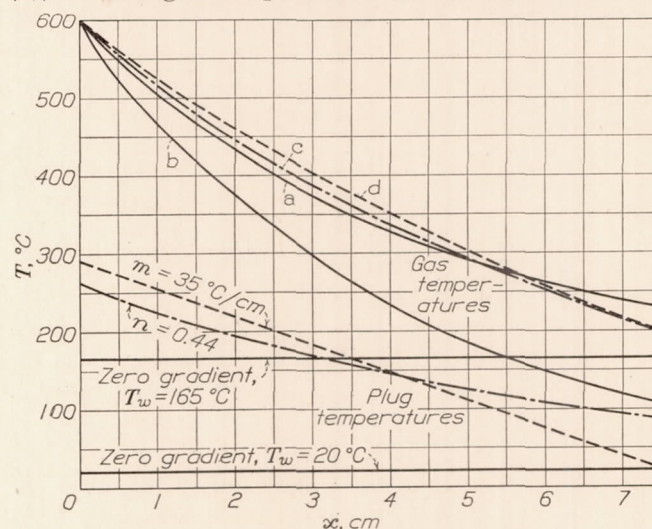


FIGURE 7.—Gas temperatures corresponding to various forms of temperature gradient in the wall

value is considerably higher than that for the final 3-inch model and may indicate why the pack with larger passages failed to operate properly.

The measured value of the end temperature of the 1-inch plug at equilibrium was about 175°C . (See p. 5.) The exit gas temperature for the 1-inch plug may be taken from curve a at $x = 2.5 \text{ cm}$. This value is about 400°C . In view of the performance of the 1-inch pack, it may be inferred that the temperature at which the mixture in the intake system can be ignited is in the vicinity of 400°C ., while the results with the final 3-inch model show that an exit gas temperature of about 250°C . is certainly quite safe.

Returning now to the original 3-inch plug, the condition of uniform wall temperature is realized only during the first few back fires, when the plug is at room temperature throughout. The cooling effect is then, of course, considerably greater than for the case $T_w = 165^\circ \text{C}$. For comparison, the gas temperatures corresponding to $T_w = 20^\circ \text{C}$. are given by curve b in Figure 7.

Wall temperature a fraction of gas temperature.—For a plug of finite conductivity, there will be in general a temperature gradient in the direction of the axis of the plug when equilibrium is attained; the magnitude of this gradient will depend on the conductivity of the pack material. A particularly simple assumption for the wall temperature T_w , and one which is likely to represent actual conditions satisfactorily, is that its value at any point is a given fraction of the excess of gas temperature over room temperature, i. e.—

$$T_w = n (T - T_o) \text{ where } 1 > n > 0.$$

This value of T_w substituted in (1) gives, after integration and reduction,

$$T = \left(T_i + \frac{n}{1-n} T_o \right) e^{-\frac{1-n}{NV}x} - \frac{n}{1-n} T_o \quad (6)$$

or for the numerical values assumed here,

$$T = 20 \left[\left(30 + \frac{n}{1-n} \right) e^{-\frac{1-n}{4}x} - \frac{n}{1-n} \right] \quad (7)$$

It is found that $n = 0.44$ gives a plug temperature of about 90° C. at the exit end, which is approximately the value measured in the present experiments. In Figure 7, curve c shows the gas temperature for this case. The plug temperature curve ($n = 0.44$) is also given. The average value of the plug temperature T_a is about 165° C. (330° F.); in fact the reason for choosing the value $T_w = 165^\circ \text{ C.}$ in the first calculation above was to make the average temperatures the same in the two cases.

Linear wall-temperature gradient.—Changing the value of n must alter the value of the average plug temperature. In order to investigate the effect of other values of the gradient while preserving—as a reasonable basis for comparison—the same value of the average temperature, another case is considered, in which *linear* gradients are assumed. This should also provide a good representation of actual conditions. Here we must take

$$\frac{T_w - T_a}{x - h} = -m$$

where h is half the axial length of the plug and m is the numerical value of the temperature gradient in Centigrade degrees per centimeter. This value of T_w defined above, when used in (1) gives after integration and reduction

$$T = T_a + (T_i - T_a) e^{-\frac{x}{NV}} + m[(NV + h)(1 - e^{-\frac{x}{NV}}) - x] \quad (8)$$

or, with the numerical values employed here,

$$T = 165 + 435e^{-0.25x} + m[8(1 - e^{-0.25x}) - x] \quad (9)$$

Choosing a value $m = 35^\circ \text{ C. cm}^{-1}$ makes the temperature of the cold end of the plug about 20° C. ; hence

this is the steepest gradient which may be assumed. The corresponding gas temperatures are given by curve d of Figure 7.

It is noted at once that the three gas temperature curves, corresponding to widely different plug temperature gradients—hence to different values of the conductivity—lie very close together. There results the rather interesting conclusion that when equilibrium has been attained, *the conductivity of the plate material is of slight importance in determining the rate of cooling of the gas.*

DISCUSSION

The preceding considerations indicate the principal conditions which must be satisfied in order to insure the efficient and positive action of a flame arrestor of the type investigated.

Of prime importance is the l/D ratio of the individual channels, which must exceed a certain minimum value in order that the gases be sufficiently cooled. The exit temperature of the gas as given, for example, by equation (4)—using the value of NV from (3)—may be written

$$T_e = T_w + (T_i - T_w) e^{-k \frac{l}{D}}$$

which shows that $\frac{l}{D}$ should be large if a low value of T_e , the exit gas temperature, is to be attained. An upper limit for this ratio is set by conditions of a practical nature, especially by the necessity of keeping the air resistance of the plug down to a permissible value, and also in order to avoid a plug of excessive size and weight.

As shown by the calculation on page 6, the rise in temperature of the plug per explosion is very small. The only effect of the heat capacity of the arrestor is to determine the rate at which the equilibrium temperature is approached. The use of a plug of large thermal capacity will therefore delay, but will not in itself prevent, back-firing. But the flame arrestor must be able to withstand an indefinite number of explosions, so that the magnitude of the thermal capacity of the plug is of no concern.

The analysis shows that the thermal conductivity of the material of which the plug is made is also of little consequence in determining the effectiveness of the device. The plates should, of course, always be made of metal, in order to insure adequate heat transfer from the gas to the channel walls. Some of the corrosion-resisting steel alloys now available would be well adapted to this use.

CONCLUSIONS

It is concluded that a flame arrestor of the type described can be made adequate to insure protection against back fires at the expense of some loss of volumetric efficiency. This loss can be made small by employing plugs of large cross section.

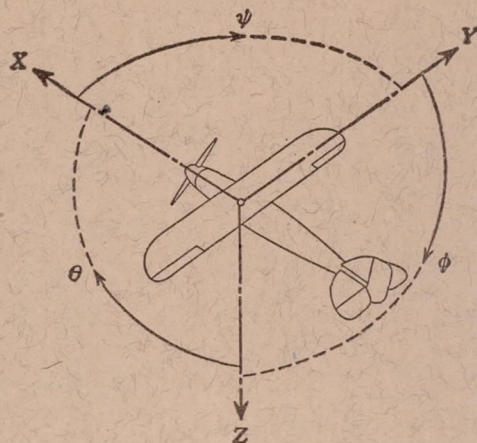
The factor of first importance in determining the effectiveness of the arrestor is the l/D ratio of the individual passages.

The mass of the plug and the thermal conductivity of the material of which it is made are of secondary importance.

LANGLEY MEMORIAL AERONAUTICAL LABORATORY,
NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
LANGLEY FIELD, VA., *October 13, 1931.*

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Positive directions of axes and angles (forces and moments) are shown by arrows

Axis			Moment about axis			Angle		Velocities	
Designation	Sym- bol	Force (parallel to axis) symbol	Designation	Sym- bol	Positive direction	Designa- tion	Sym- bol	Linear (compo- nent along axis)	Angular
Longitudinal---	X	X	rolling-----	L	Y → Z	roll-----	φ	u	p
Lateral-----	Y	Y	pitching-----	M	Z → X	pitch-----	θ	v	q
Normal-----	Z	Z	yawing-----	N	X → Y	yaw-----	ψ	w	r

Absolute coefficients of moment

$$C_l = \frac{L}{qbS}$$

$$C_m = \frac{M}{qcS}$$

$$C_n = \frac{N}{qbS}$$

Angle of set of control surface (relative to neu-
tral position), δ . (Indicate surface by proper
subscript.)

4. PROPELLER SYMBOLS

D , Diameter.

p , Geometric pitch.

p/D , Pitch ratio.

V' , Inflow velocity.

V_s , Slipstream velocity.

T , Thrust, absolute coefficient $C_T = \frac{T}{\rho n^2 D^4}$

Q , Torque, absolute coefficient $C_Q = \frac{Q}{\rho n^2 D^5}$

P , Power, absolute coefficient $C_P = \frac{P}{\rho n^3 D^5}$.

C_s , Speed power coefficient $= \sqrt[5]{\frac{\rho V^5}{P n^2}}$.

η , Efficiency.

n , Revolutions per second, r. p. s.

Φ , Effective helix angle $= \tan^{-1} \left(\frac{V}{2\pi r n} \right)$

5. NUMERICAL RELATIONS

1 hp = 76.04 kg/m/s = 550 lb./ft./sec.

1 kg/m/s = 0.01315 hp

1 mi./hr. = 0.44704 m/s

1 m/s = 2.23693 mi./hr.

1 lb. = 0.4535924277 kg.

1 kg = 2.2046224 lb.

1 mi. = 1609.35 m = 5280 ft.

1 m = 3.2808333 ft.